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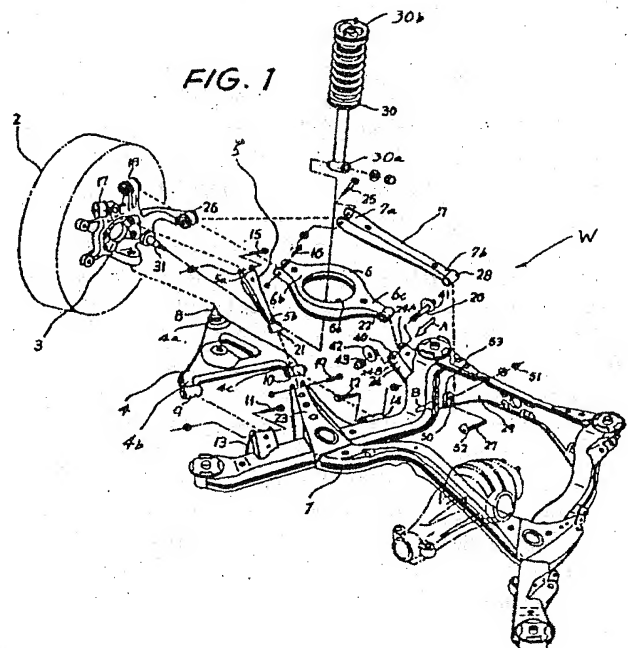
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54 **Wheel alignment adjusting mechanism.**

57 A wheel alignment adjusting mechanism for a rear suspension system for an automotive vehicle. The wheel alignment adjusting mechanism is comprised of a rear upper link for movably connecting a wheel supporting member and a bracket projecting from a suspension member assembly. The rear upper link is pivotally connected at one end thereof with the bracket through a pivot pin. The pivot pin is movably fitted within elongate openings which are formed respectively in two side plate sections of the brackets. The longitudinal axis of each elongate opening inclines to be lowered in the outboard direction by an angle of 30 degrees relative to a horizontal plane in a vehicle body, thereby maintaining smaller change amounts in wheel alignment along with suspension stroke.



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WHEEL ALIGNMENT ADJUSTING MECHANISM

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to improvements in a mechanism for adjusting wheel alignment of a vehicle, and more particularly to a device for maintaining smaller the change in wheel alignment even upon suspension stroke.

2. Description of the Prior Art

Wheel alignment of a vehicle largely influences vehicle cruising characteristics and driveability and therefore is important from the view points of safety and ride-in comfortableness of the vehicle. Usually vehicles are so arranged that wheel alignment is to be adjustable by a mechanism even after assembly thereof. An example of such a wheel alignment adjusting mechanism is disclosed in Japanese Utility Model Provisional Publication No. 56-24371, in which a pivot pin of a suspension link can be shifted within an elongate opening formed in a member on the side of a suspension member in order to adjust toe angle, camber angle and the like.

However, drawbacks have been encountered in such a conventional wheel alignment adjusting mechanism, in which toe angle change (the change amount in wheel alignment) along with suspension stroke (bound and rebound of road wheels) becomes larger and therefore the toe angle becomes far from a design standard (represented by the character d) as seen from the characteristics represented by characters a_0 , a_1 in Fig. 7. The toe angle characteristics a_0 , a_1 in Fig. 7 are obtained by using a double wish-bone type rear suspension system in which a pivot pin for the inboard end section of a rear upper link was displaced inboard and outboard by 2 mm within the elongate opening extending in a direction from the inboard to outboard sides of a vehicle body. This means that the vehicle cruising characteristics and driveability corresponding to an initial wheel alignment cannot be maintained and unavoidably changes, thereby causing a driver to get disoriented.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide an improved wheel alignment adjusting mechanism which is less in amount of change in wheel alignment even upon suspension stroke, maintaining stable cruising characteristics and driveability.

The wheel alignment adjusting mechanism according to the present invention comprises a suspension link through which a road wheel and a suspension member connected to a vehicle body are movably connected with each other. An elongate opening is formed on the side of the suspension member and extends generally in the direction of inboard to outboard sides of the vehicle body. The longitudinal axis of the elongate opening inclines relative to a horizontal plane in the vehicle body so that the amount of change in wheel alignment along with suspension stroke is maintained smaller. The suspension link is pivotally connected at its one end with a pivot pin which is disposed within the elongate opening and movable in the longitudinal direction of the elongate opening.

Accordingly, dislocating the pivot pin for the suspension link within the elongate opening causes the road wheel to be displaced in a corresponding amount, thereby adjusting wheel alignment thus to decide an initial wheel alignment. Moreover, by virtue of inclination of the longitudinal axis of the elongate opening, the change amount of wheel alignment along with suspension stroke can be maintained smaller regardless of locations of the pivot pin within the elongate opening. This can maintain the vehicle cruising characteristics and driveability corresponding to the initial wheel alignment even if vehicle occupants increase or decrease and/or during a vehicle cruising with elastic bound and rebound of the road wheel, thereby preventing a driver from getting disoriented.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is an exploded perspective view of an essential part of a rear suspension system for a vehicle, provided with an embodiment of a wheel alignment adjusting mechanism in accordance with the present invention;

Fig. 2 is a side view of an essential part of the wheel alignment adjusting mechanism of Fig. 1, as viewed from the direction of an arrow A of Fig. 1;

Fig. 3 is a cross-sectional view taken in the direction of arrows substantially along the line III-III of Fig. 2;

Fig. 4 is a side view of another essential part of the wheel alignment adjusting mechanism of Fig. 1, as viewed from the direction of an arrow B of Fig. 1;

Fig. 5 is a cross-sectional view taken in the direction of arrows substantially along the line V-V of Fig. 4;

Fig. 6 is a graph showing the respective change amounts in wheel alignment upon dislocation of various suspension links of the suspension system of Fig. 1;

Fig. 7 is a graph showing the respective change amounts in wheel alignment in cases of various inclination angles of the longitudinal direction of an elongate opening in which a rear upper link pivot pin of the suspension system in Fig. 1 is disposed; and

Fig. 8 is a graph showing the camber change amount and the roll steer rate upon dislocation of the rear upper link pivot pin of Fig. 7, in terms of the inclination angle of the longitudinal axis of the elongate opening.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to Fig. 1, there is shown an embodiment of a wheel alignment adjusting mechanism W in accordance with the present invention. In this embodiment, the wheel alignment adjusting mechanism W is incorporated in a rear suspension system for an automotive vehicle. The rear suspension system includes a suspension member assembly 1 which is secured to a vehicle body (not shown) of the automotive vehicle in a vibration insulating manner. A rear right wheel 2 is rotatably supported by a wheel supporting member 3 which is movably connected to the suspension member assembly 1.

A A-shaped arm 4, a front upper link 5, a rear upper link 6 and a lower link 7 are movably disposed in a manner to connect the wheel supporting member 3 and the suspension member assembly 1. The A-shaped arm 4 has an inboard tip corner section 4a which is connected through a ball joint 8 with the lower section of the wheel supporting member 3. The A-shaped arm 4 has two projecting outboard base end sections 4b, 4c which are respectively provided with elastic bushings 9, 10. The elastic bushings 9, 10 are respectively mounted on pivot pins 11, 12 which are respectively fixedly secured to brackets 13, 14 of the suspension member assembly 1. Accordingly, the projecting base end sections 4b, 4c are movably connected to the brackets 13, 14, respectively, in such

a manner that the A-shaped member 4 is vertically movable relative to the vehicle body. The front upper link 5 has an outboard end section 5a which is provided with a pivot pin 15 which is installed in a bushing 17 secured at an upper section of the wheel supporting member 3 so that the front upper link 5 is pivotal relative to the wheel supporting member 3. The inboard end section 5b of the front upper link 5 is provided with an elastic bushing 21 mounted on a pivot pin 19 fixedly secured to a bracket 23 of the suspension member assembly 1 so that the front upper link 5 is vertically pivotally movable relative to the suspension member assembly 1. The rear upper link 6 is shaped generally annular to form an opening 6a and has an outboard end section 6b provided with a pivot pin 16. The pivot pin 16 is disposed in an elastic bushing 18 securely supported at an upper section of the wheel supporting member 3 so that the rear upper link 6 is vertically pivotally movable relative to the wheel supporting member 3. The inboard end section 6c of the rear upper link 6 is provided with an elastic bushing 22 mounted on a pivot pin 20 secured to the bracket 24 of the suspension member assembly 1 so that the rear upper link 6 is vertically pivotally movable relative to the suspension member assembly 1.

The lower link 7 has an outboard end section 7a which is provided with a pivot pin 25 which is inserted in an elastic bushing 26 secured at the rear and lower section of the wheel supporting member 3 so that the lower link 7 is vertically pivotally movable relative to the wheel supporting member 3. The inboard end section 7b of the lower link 7 is provided with an elastic bushing 28 which is mounted on a pivot pin 27 secured to a crossbar 29 of the suspension member assembly 1 in such a manner that the lower link 7 is vertically pivotally movable relative to the crossbar 29 of the suspension member assembly 1.

A strut or shock absorber 30 is disposed piercing the opening 6a of the rear upper link 6 and vertically extends. The strut 30 has a lower end section 30a pivotally connected to a laterally inboard extending shaft 31 of the wheel supporting member 3. The upper end section 30b of the strut 30 is connected to the vehicle body through a strut mount insulator (not shown).

With the thus arranged rear suspension system, toe angle and camber angle alter as shown in Fig. 6 if the pivot pins of the suspension links are displaced inboard and outboard in the vehicle body. In Fig. 6, the characters e_0 , e_1 indicate wheel alignment change amounts in case the pivot pin 11 of the A-shaped arm 3 is displaced outboard and inboard by 2 mm in the vehicle body, respectively. The characters f_0 , f_1 indicate wheel alignment change amounts in case the pivot pin 19 of the

front upper link 5 is displaced outboard and inboard by 2 mm in the vehicle body, respectively. The characters g_o , g_i indicate wheel alignment change amounts in case the pivot pin 20 of the rear upper link 6 is displaced outboard and inboard by 2 mm in the vehicle body, respectively. The characters h_o , h_i indicate alignment change amounts in case the pivot pin 27 of the lower link 7 is displaced outboard and inboard by 2 mm in the vehicle body, respectively.

As is apparent from Fig. 6, in order to adjust camber angle, it is preferable to displace the pivot pin 20 of the rear upper link 6 in the inboard and outboard directions of the vehicle because of the minimum influence to other alignments. Additionally, in order to adjust toe angle, it is preferable to displace the pivot pin 27 of the lower link 7 in the inboard and outboard directions of the vehicle body because of the minimum influence to other alignments. In view of the above, in this embodiment, a chamber adjusting mechanism is employed in connection with the pivot pin 20 of the rear upper link 6 and arranged as shown in Figs. 2 and 3, while a toe adjusting mechanism is employed in connection with the pivot pin 27 of the lower link 7 and arranged as shown in Figs. 4 and 5.

The chamber adjusting mechanism will be discussed with reference to Figs. 2 and 3.

The bracket 24 includes two opposite side plate sections 24A, 24B which are spaced from each other and parallel with an imaginary vertical plane (not shown) to which a longitudinal axis of the vehicle body is perpendicular. In this connection, the crossbar 29 extends parallel with the imaginary vertical plane. The side plate sections 24A, 24B are respectively formed with elongate openings 40, 40 which are identical in shape and location in the side plate sections. The pivot pin 20 are disposed piercing the elongate openings 40, 40 of the side plate sections 24A, 24B so as to be adjustably secured to the bracket 24. The elastic bushing 22 at the inboard end section of the rear upper link 6 is mounted on the pivot pin 20 as shown in detail in Fig. 3. The elastic bushing 22 includes a cylindrical elastomeric member 22a tightly interposed between an inner cylindrical member 22b and an outer cylindrical section 22c forming part of the inboard end section 6c of the rear upper link 6. The inner cylindrical member 22b is slidably mounted on the pivot pin 20.

A disc plate 41 having an eccentric opening is disposed between the hexagonal head 20a of the pivot pin 20 and the side plate section 24A of the bracket 24 under the state in which the pivot pin 20 pierces the elongate openings 40 of the side plate sections 24A, 24B of the bracket 24 and the eccentric opening of the disc plate 41. The eccentric opening of the disc plate 41 is located eccentric

relative to the disc plate 41. The disc plate 41 is in tight contact with the pivot pin head 20a and the side plate section 24A of the bracket 24. Another disc plate 42 has an eccentric opening and identical in shape with the disc plate 42. The disc plate 42 is interposed between a nut 43 engaged on the end section opposite to the head 20a under the state the pivot pin 20 pierces the elongate openings 40 of the side plate sections 24A, 24B of the bracket 24 and the eccentric opening of the disc plate 42. The disc plate 42 is in tight contact with the side plate section 24B and the disc plate 42. It will be understood that the nut 43 engaged with the pivot pin 20 functions to prevent the pivot pin 20 from getting out of the bracket 24 and additionally the disc plate 42 from getting off.

The disc plate 41 is movably put between a flange 24a and a projecting portion 24c of the side plate section 24A. The flange 24a extends in the longitudinal direction of the side plate section 24A and projects outwardly from the side plate section 24A. The flange 24a is formed by bending the lower edge portion of the side plate section 24A. The projecting portion 24c is formed integral with the side plate section 24A. Additionally, the side plate section 24B is formed with a flange 24b and a projecting portion 24d which respectively correspond to the flange 24a and the projecting portion 24c in the side plate section 24A. Accordingly, the disc plate 42 is movably put between the flange 24b and the projecting portion 24d.

The elongate opening 40 of the side plate section 24A of the bracket 24 is arranged such that the longitudinal axis L thereof inclines an angle θ relative to a horizontal plane P, on a vertical plane containing the longitudinal axis L. The horizontal plane is horizontal relative to the vehicle body and generally parallel with a floor (not shown) of the vehicle body. Additionally, the longitudinal axis L of the elongate opening 40 is lowered relative to the horizontal plane P in a direction from the inboard side to the outboard side of the vehicle body as clearly shown in Fig. 2. It will be understood that the longitudinal axis L of the elongate opening 40 in the side plate section 24B inclines as same as in that in the side plate section 24A. The inclination angle θ of the longitudinal axis L of the elongate opening 40 is preferably selected at a larger value to approach a design standard d with reference to the characteristics indicated by characters b_i , b_o and characters c_i , c_o in Fig. 7. In Fig. 7, the character b_i indicates a case in which the pivot pin 20 is displaced inboard by 2 mm in the elongate opening 40 having the longitudinal axis inclination angle θ of 30° , while the characters b_o indicates a case in which the pivot pin 20 is displaced outboard by 2 mm in the elongate opening 40 having the longitudinal axis inclination angle θ of 30° . The

character c_i indicates a case in which the pivot pin 20 is displaced inboard by 2 mm in the elongate opening 40 having the longitudinal axis inclination angle θ of 45° , while the character c_o indicates a case in which the pivot pin 20 is displaced outboard by 2 mm within the elongate opening 40 having the longitudinal axis inclination angle θ of 45° . For reference, the characters a_i , a_o indicate cases in which the pivot pin 20 is displaced inboard and outboard by 2 mm within the elongate opening having the longitudinal axis inclination angle (θ) of 0° , respectively. It will be understood that the cases indicated by the characters a_i , a_o are not within the scope of the present invention.

In this connection, variation in camber angle (degree) and rate of roll steer upon bound of 40 mm are shown in Fig. 8. In Fig. 8, the characters j_o , k_o and l_o represent the cases in which the pivot pin 20 is displaced outboard by 2 mm within the elongate openings 40 having the longitudinal axis inclination angle θ of 0° , 30° and 45° , respectively. The characters j_i , k_i , l_i represent the cases in which the pivot pin 20 is displaced inboard by 2 mm within the elongate openings 40 having the longitudinal axis inclination angle θ of 0° , 30° and 45° . The graph in Fig. 8 demonstrates that enlarging the longitudinal axis inclination angle θ of the elongate opening 40 reduces the roll steer rate. However if the longitudinal axis inclination angle θ is too large, a camber adjusting range is minimized even though the displacement amount (2 mm) of the pivot pin 20 is constant as seen from Fig. 8, thus lowering camber adjusting efficiency. In view of these, it will be understood that the longitudinal axis inclination angle θ of the elongate opening 40 is preferably about 30° .

Next a toe adjusting mechanism will be discussed with reference to Figs. 4 and 5.

In this embodiment, the crossbar 29 includes two opposite side plate sections 29A, 29B which are spaced from and parallel with each other. An elongate opening 50 is formed in each side plate section 29A, 29B. The pivot pin 27 passes through the elongate openings 50 of the opposite side plate sections 29A, 29B, passing through the elastic bushing 28 of the lower link 7. The longitudinal axis L_o of the elongate opening 50 is parallel with the horizontal plane P, on a vertical plane containing the longitudinal axis L_o . The elastic bushing 28 is located between the opposite side plate sections 29A, 29B and includes a cylindrical elastomeric member 28a securely disposed between an inner cylindrical member 28b and an outer cylindrical section 28c. The inner cylindrical member 28a is mounted on and in slidable contact with the pivot pin 27. The outer cylindrical section 28c forms part of the lower link 7. A disc plate 52A having an eccentric opening is disposed between the hexag-

onal head 27a of the pivot pin 27 and the side plate section 29B in such a manner as to be in tight contact with the pivot pin head 27a and the side plate section 29B, in which the pivot pin 27 passes through the eccentric opening of the disc plate 52A. The eccentric opening is located eccentric in the disc plate 52A. Another disc plate 52B having an eccentric hole is identical with the disc plate 52A and disposed between the side plate section 29A and a nut 51 engaged with the end section (opposite to the head 27a) of the pivot pin 27. The disc plate 52B is in tight contact with the side plate section 29A and with the nut 51. Each of the disc plates 52A, 52B slidably fits in a vertically extending groove 53 formed in the side plate sections 29A, 29B in such a manner that the periphery of the disc plate 52A, 52B in slidable contact with the side walls 53a, 53a of the groove 53 so that the disc plate can be guided between the side walls 53a, 53a. The longitudinal axis of the grooves 53 is perpendicular to the longitudinal axis L_o of the elongate opening 50. It will be understood that the longitudinal axis L_o may incline relative to the horizontal plane P.

The manner of operation of the thus arranged wheel alignment adjusting mechanism W will be discussed hereinafter.

In adjusting camber in the wheel alignment, first the pivot pin 20 is rotated camber in the wheel alignment, the pivot pin 20 is rotated by rotating the hexagonal head 20a upon loosening the nut 43. This rotates the disc plate 41, 42 together with the pivot pin 20, in which the disc plate 41, 42 moves upon being guided between the flange 24a, 24b and the projecting portion 24c, 24d so that the pivot pin 20 displaces along the length of the elongate opening 40. Accordingly, the rear upper link 6 is longitudinally displaced. Additionally, since the longitudinal axis inclination angle θ is 30° , camber adjustment can be achieved depending upon the displacement amount of the pivot pin 20 and according to the characteristics shown in Fig. 8.

Now, as appreciated from Figs. 7 and 8, inclining the longitudinal axis of the elongate opening 40 by an angle (O) of 30° minimizes toe angle change (roll steer rate) caused by bound and rebound, as compared with the characteristics of the cases (represented by the characters, j_i , j_o) in which the elongate opening (40) has no inclination of the longitudinal axis. Accordingly, vehicle cruising characteristics and drivability corresponding to an initially set wheel alignment can be maintained even if vehicle passengers are reduced and/or even during vehicle cruising upon elastic bound of road wheels accompanying suspension stroke. This prevents a driver from getting disoriented while improving safety in vehicle driving.

In adjusting toe of the wheel alignment, first the pivot pin 27 is rotated by rotating the hexagonal head 27a upon loosening the nut 51. This causes the disc plates 52A, 52B to rotate together with the pivot pin 27, in which the disc plate 52A, 52B moves upon being guided between the side walls 53a, 53b of the groove 53. Accordingly, the pivot pin 27 is moved along the length of the elongate opening 50, and therefore the lower link 7 displaces longitudinally thereby adjusting toe angle of the wheel 2.

Thus in this embodiment, since camber adjustment is accomplished by displacing the upper link pivot pin 20 while toe adjustment is accomplished by displacing the lower link pivot pin 27, these adjustments hardly affect each other in alignment adjustment as apparent from Fig. 6. This provides the same effect as in a wheel alignment adjusting manner in which camber adjustment and toe adjustment are independently carried out, while facilitating wheel alignment adjusting operation.

In this embodiment, guiding the disc plates 41, 42, 52A and 52B is accomplished by the machined portions 24a, 24b, 24c, 24d; 53a, 53b of the conventional parts without requiring special separate parts. This lowers the production cost of the wheel alignment adjusting mechanism while facilitating to improve the accuracy of the guiding portions for the disc plates.

While the principle of the present invention has been shown and described as being applied only to the suspension system in which an upper link system is constituted of the front and rear upper links 5, 6, it will be understood that the principle may not be limited to be applied to it and therefore applied to suspension systems of the so-called double wish-bone type wherein, for example, a A-shaped arm is used as an upper link and to one of the so-called strut type such as parallel link type. The principle of the present invention is particularly effective if wheel alignment is adjusted with the A-shaped arm of the suspension system of the double wish-bone type in which there is greater influence of suspension stroke (suspension movement due to bound and rebound of road wheels) to wheel alignment.

Claims

1. A wheel alignment adjusting mechanism for a vehicle, comprising:
a suspension link for movably connecting a road wheel and a suspension member connected to a vehicle body;
means defining an elongate opening extending generally in direction of inboard to outboard sides of the vehicle body, said elongate opening being

located on side of said suspension member;
a pivot pin disposed within said elongate opening and movable in longitudinal direction of said elongate opening, said suspension link having a first end section pivotally connected to said pivot pin; and

means inclining longitudinal axis of said elongate opening relative to a horizontal plane in the vehicle body so as to reduce amount of change in wheel alignment along with suspension stroke.

2. A wheel alignment adjusting mechanism as claimed in Claim 1, wherein said suspension link connects a wheel supporting member for rotatably supporting said road wheel and a bracket secured to said suspension member.

3. A wheel alignment adjusting mechanism as claimed in Claim 2, wherein said elongate opening is formed in said bracket.

4. A wheel alignment adjusting mechanism as claimed in Claim 1, wherein the longitudinal axis of said elongate opening is lowered in an outboard direction relative to the horizontal plane.

5. A wheel alignment adjusting mechanism as claimed in Claim 4, wherein the longitudinal axis of said elongate opening inclines by an angle of about 30 degrees relative to the horizontal plane.

6. A wheel alignment adjusting mechanism as claimed in Claim 1, wherein said first end section of said suspension link has an elastic bushing mounted on said pivot pin.

7. A wheel alignment adjusting mechanism as claimed in Claim 6, wherein said suspension link has a second end section which is connected through an elastic bushing with said wheel supporting member.

8. A wheel alignment adjusting mechanism as claimed in Claim 2, wherein said bracket includes first and second side plate sections which are parallel with each other and extend parallel with a vertical plane, in the vehicle body, to which a longitudinal axis of the vehicle body is perpendicular, each side plate section being formed with said elongate opening, said pivot pin passing through said elongate openings of said first and second side plate sections, said suspension link first end section being located between said first and second side plate sections.

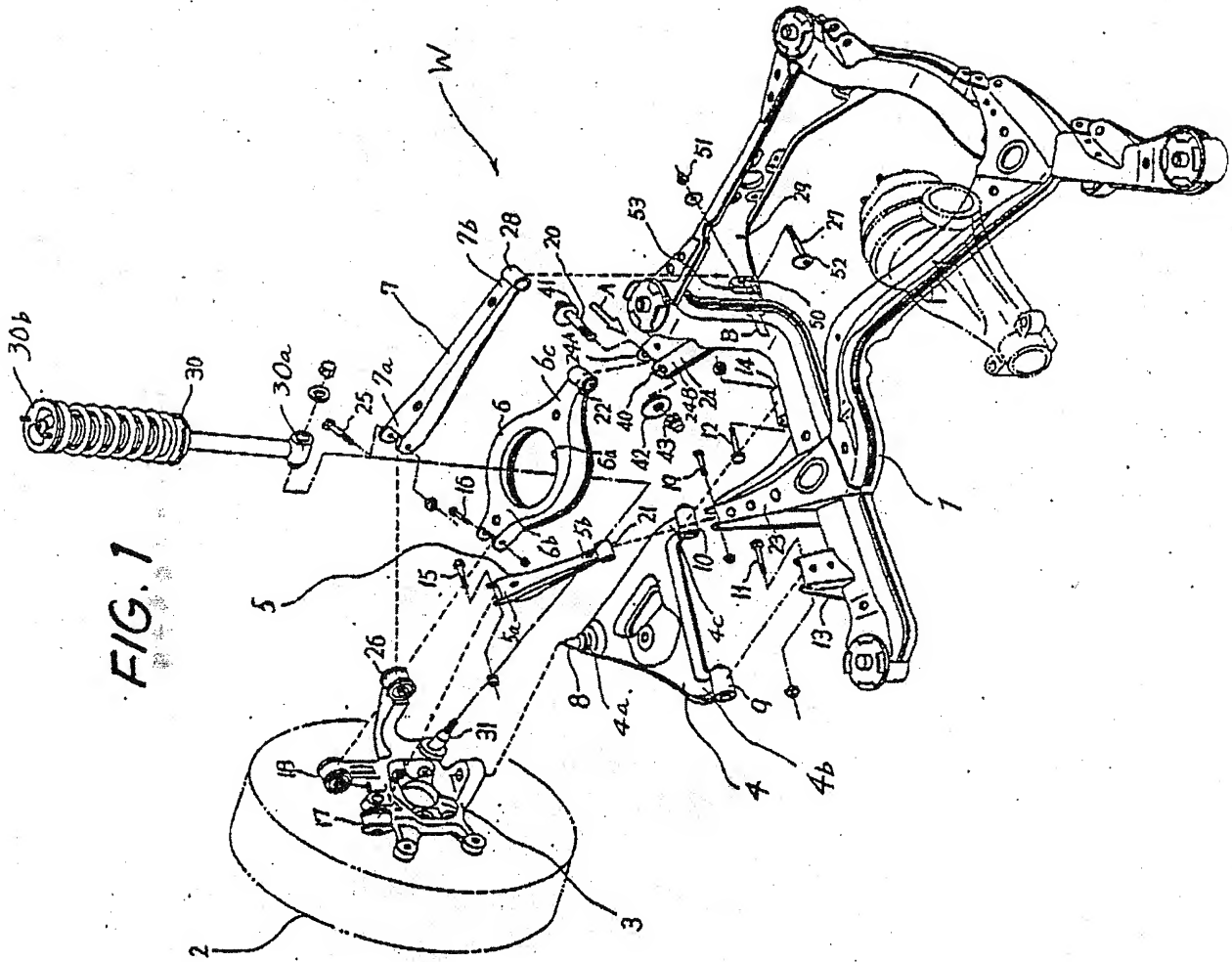


FIG. 2

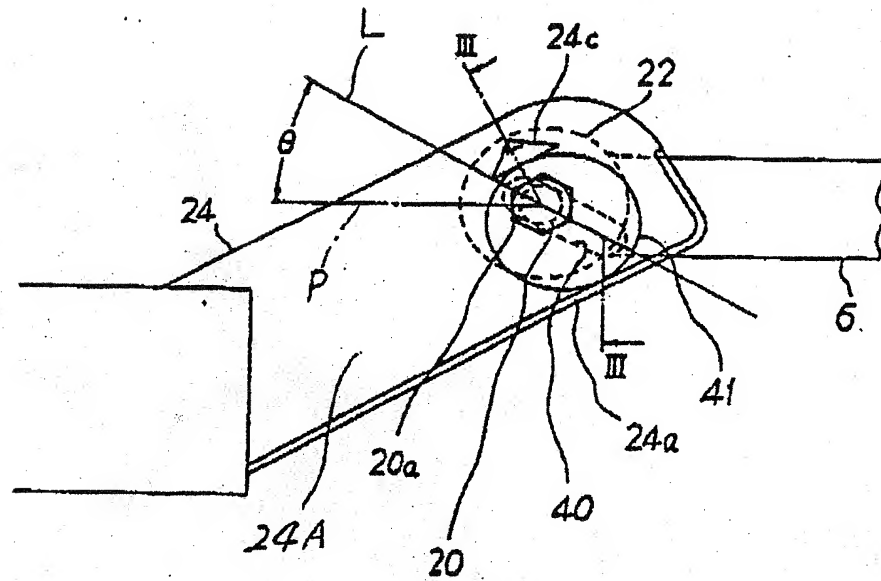


FIG. 3

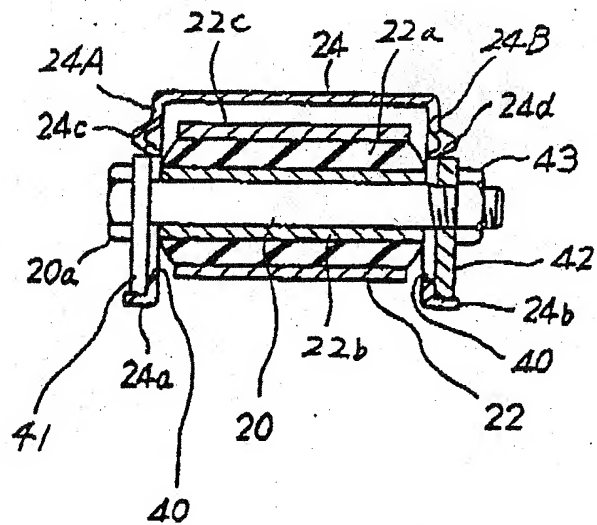


FIG. 4

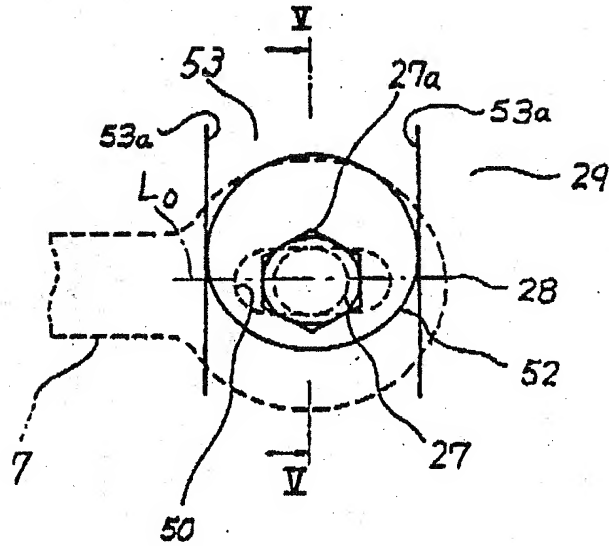


FIG. 5

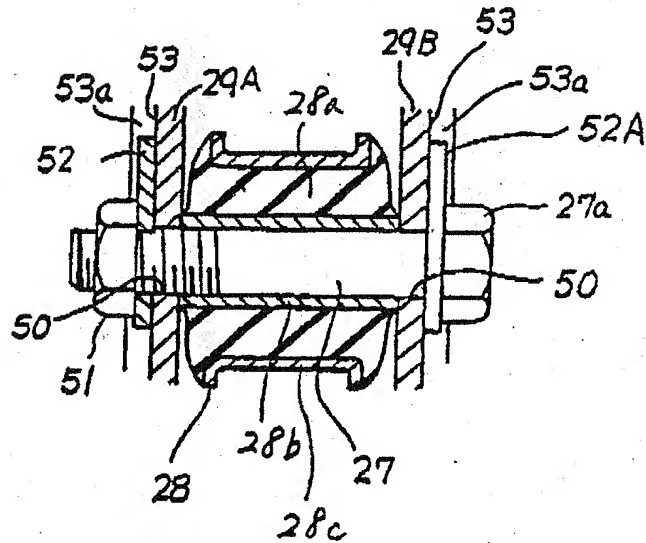


FIG. 6

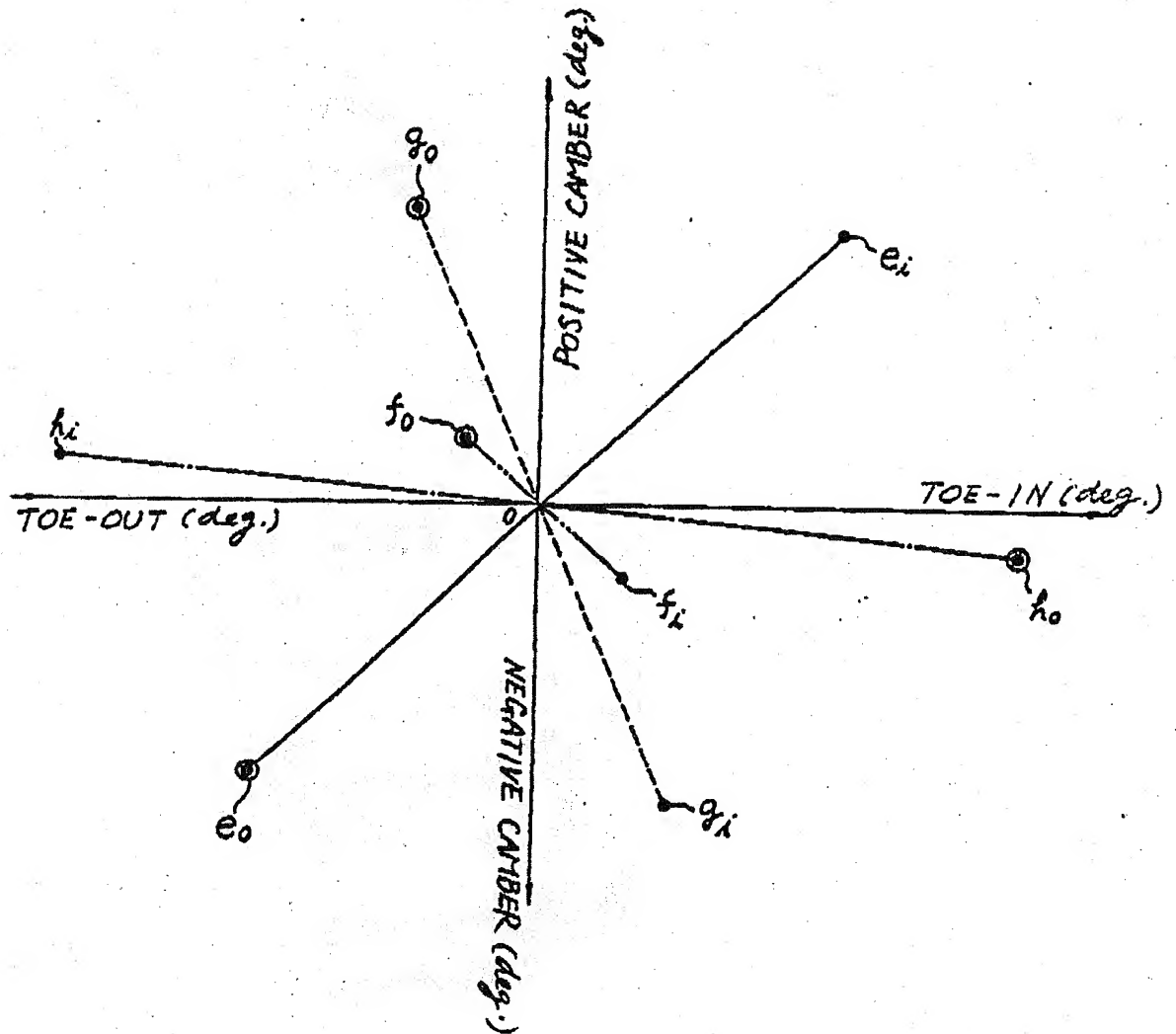


FIG. 7

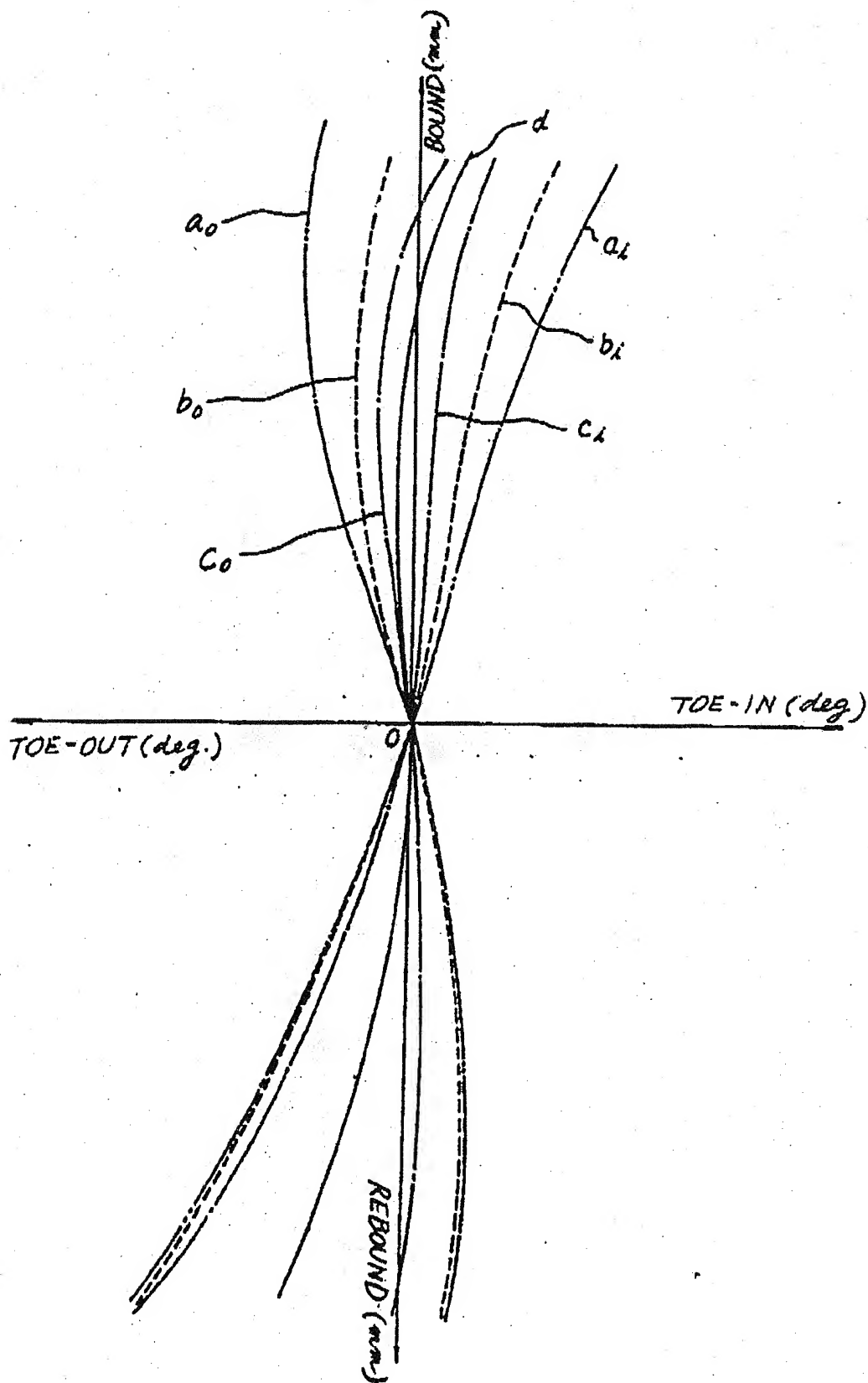


FIG. 8

